



Initial Experiments of High-Speed Drive System Windage Losses

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Abstract

High speed gearing performance is very important to the overall drive system efficiency. Certain losses such as gear meshing and bearing drag can be minimized by design changes such as pressure angle of the gears and the geometry and type of bearings being used. One component that can have a large effect on the overall performance of high-speed drive systems is the parasitic drag known as gear windage. This loss mechanism is not well understood and minimizing this component is usually accomplished through much trial and error. The results presented in this paper will document some of the design parameter effects on the amount of windage losses. A new test facility at NASA Glenn has been assembled to systematically study the design variables. Results from recent tests will be presented. The tests are for a single gear, with and without lubricants, and some initial studies using shrouds.

1.0 Introduction

Power loss in high-speed gear systems due to windage can be a substantial portion of the total losses of the drive system. In the open literature this topic has not been investigated to the levels of other important gear research topics. However, in high speed gearing this loss mechanism is very important.

When pitch line velocity is above 125 m/s (25000 ft/min), windage losses can be half or more of the total losses from the drive system. The rest of the losses can be summarized as that due to the gear meshing, bearings, and seals in the drive system under study (Refs. 1 and 2).

In prior work (Ref. 3) the gear windage effect on drive system power loss was extrapolated from work done on gas turbine disk power loss. Using this approach, the results were found from a few simple calculations and were dependent on the diameter and rotational speed. For a spur gear, the losses were found to be related to the gear size to the 3rd power and rotational speed to the 5th power.

In the studies of Dawson (Refs. 4 and 5), a more gear-like representation was provided. Many typical gear features were varied such as gear helix angle, diameter to face width ratio, enclosure factor, and environmental quantities (viscosity and density). Power loss was found by the decelerated of the system. The data was then used to generate a calculation procedure for a single gear. A further refinement on the dependencies of gears and rotational speed were found.

Winfree (Ref. 6) presented windage issues for intersecting axis gearing (spiral bevel). Gears were operated in a manner that the windage driven fluid could be observed. In this study, shrouding location, clearance, and shroud exit locations were investigated. Also, the power to drive the system was measured and therefore the most efficient arrangements were found. Shrouds on the gears helped reduce the losses on the order of 70% on test run to 25,000 ft/min (127 m/s). Also it was found that the closer the shrouding the lower the power loss.

The work of Diab et al. (Ref. 7) also investigated the windage losses of a single gear that was rotated in an ambient (air) environment. Their data was then compared to prior methods and their own formulation to predict the windage loss behavior.

In some recent work (Refs. 8 to 12), analytical and experimental studies on gear windage have been accomplished. The use of computational fluid dynamic (CFD) methods has begun to be applied to this very difficult problem. Not only is the modelling difficult, especially through the meshing zone, but also many other aspects such as two-phase flow and shrouding can further complicate the process to provide and accurate assessment of the power losses.

The objective of this initial study was to investigate the windage behavior of a single, fairly large spur gear. Tests were conducted from low to very high rotational speed—where windage losses can be the dominant source. Tests were conducted with and without lubricant present, and/or using shrouding. The data that has been generated in this study can be used for analytical model validation or empirical model verification for very high-speed conditions.

2.0 Test Facility

The test facility used in this study is the newly operational Gear Windage Test Facility at NASA Glenn Research Center. A sketch of the test facility is shown in Figure 1. The facility has a 112 kW (150 hp) DC drive motor that is connected to a 5.7:1 speed up gearbox. The output of the speed-up gearbox then is connected to a torque meter prior to coupling connection of the input shaft to the test gearbox. The input and output shafts have hydraulically operated clutches that allow the facility (in single or dual shaft mode) to be disconnected from the power applier and/or magnetic brake attached to the output shaft. With the speed capability of the drive motor and speed increasing gearbox and the dimensions of the test specimen, the pitch line velocity can be taken to 280 m/s (~55,000 ft/min).

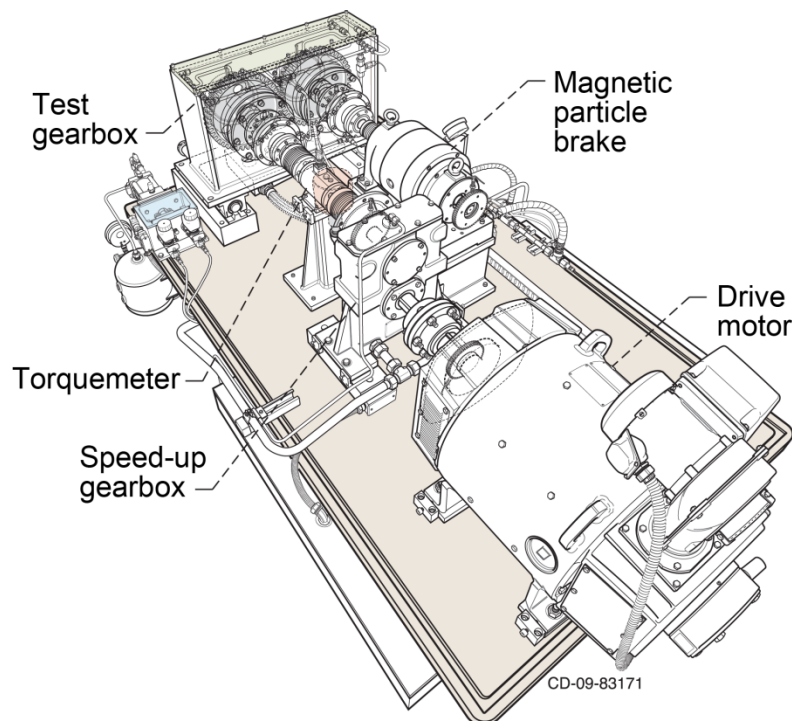


Figure 1.—Overview sketch of the NASA Glenn Research Center Gear Windage Rig.

The test gears can be run in the gearbox with or without shrouding. The shrouding clearance can be adjusted radially and axially. In the data presented in this paper, only results from a single spur gear will be discussed. The gear under test in this report is given in Table 1 and the maximum and minimum shroud clearances are provided in Table 2. A sketch of the shrouding arrangement is shown in Figure 2.

Due to the high-speed conditions of the facility, safety precautions were taken to provide safe operation at the extremely high rotational speed. A photograph of the facility with containment shield in place is shown in Figure 3.

The shroud arrangement used in this study is shown in Figures 4 to 6. In Figure 4 the shroud is shown adjusted at the minimum conditions for the gear used in this study. The outer housing provided support and kept the shroud in alignment via circumferentially machined grooves. The radial clearance was maintaining via grooves machined on the sides of the shrouds.

Table 1.—Basic Gear Dimensions

Number of teeth	52
Pitch / module, 1/in. (mm)	4 (6.35)
Face Width in. (mm)	1.12 (28.4)
Pitch Diameter, in. (mm)	13.0 (330.2)
Pressure Angle, deg.	25
Outside Diameter, in. (mm)	13.49 (342.65)

Table 2.—Shroud to Gear Clearances Used in Testing

	Minimum	Maximum
Radial, in. (mm)	0.034 (0.864)	0.656 (16.66)
Axial, in. (mm)	0.05 (1.27)	1.19 (30.23)

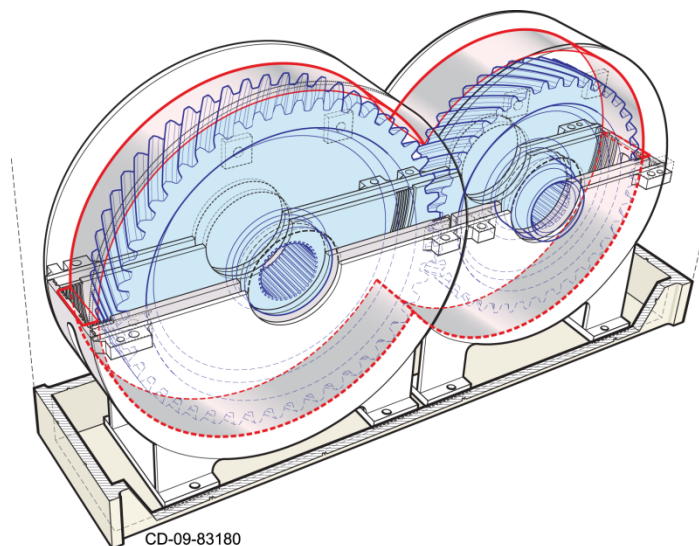


Figure 2.—Shrouding assembly for the test facility.

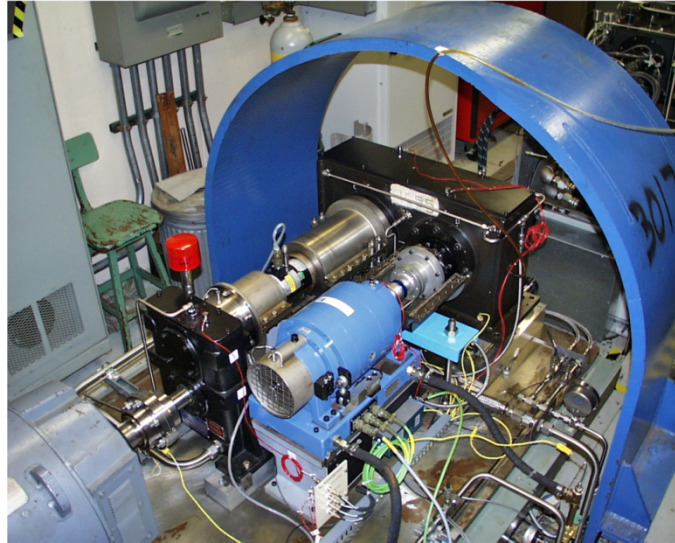


Figure 3.—Photograph of the test gearbox with containment shield in place.

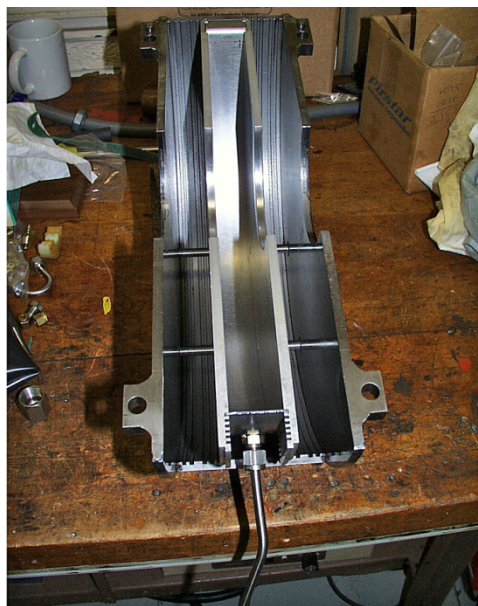


Figure 4.—Shroud before installation in test facility, showing adjustability of clearance radially and axially.

Figure 5 shows the support and bottom shroud fastened inside the test gearbox at the minimum radial and axial conditions for the gear used in this study. Figure 6 shows the entire shroud arrangement for a single gear with a block off plate for the instrumentation and lubrication jet. If there were another gear meshing, then the meshing zone would be in this region occupied by the block-off plate.

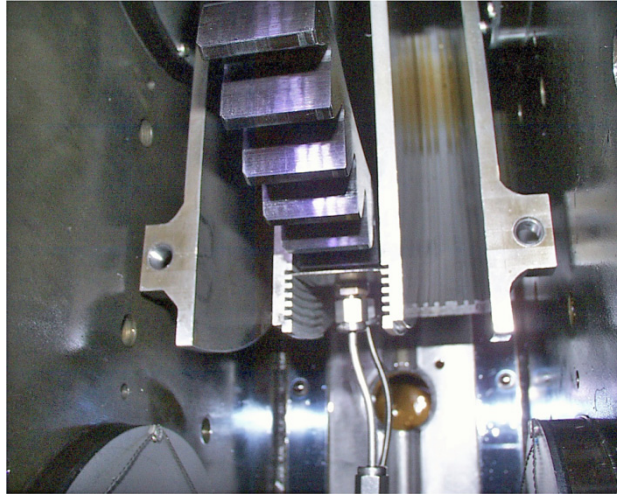


Figure 5.—Bottom shroud installed with closest axial and radial clearances.

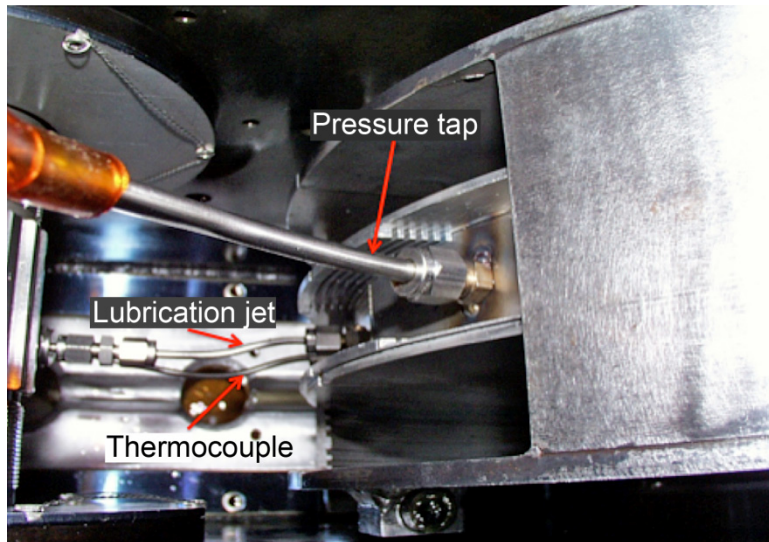


Figure 6.—Shroud installed in test facility in minimum axial and radial clearances.

3.0 Test Procedure and Data Acquisition

The facility was operated at a series of increasing drive motor speeds. At each of these conditions several different data were collected. The drive motor speed, torque applied to rotate the test hardware, internal shroud (fling-off) temperature, and internal shroud static pressure data were taken at steady drive motor speed conditions. Data was taken and then the speed was incremented up a given drive motor speed from 500 to 3125 RPM (or 2587 to 16168 RPM of the gear shaft). The data was taken from motor controller speed, a commercially available torque meter for torque, thermocouple for inside shroud temperature (fling-off), and a manometer.

4.0 Test Results

In order to determine the effects of the gear-only windage a separate test was conducted with the entire system in place minus the gear. The data from that test is shown in Figure 7. The losses from this test were mainly the support bearing and seal losses from the input shaft. Since the data to be presented in this paper were for a single gear, this test was sufficient to determine these facility support losses over the speed range of the tests to be discussed. The data from Figure 7 was used to remove these facility losses from the total losses measured when conducting tests.

4.1 Closed Shroud Results

One of the environmental-facility test conditions that was conducted included running the gear in the shroud system without a drain hole (slot) in the shrouding. This test could only be conducted in an air only environment since there would be no circumferential drain holes in the shroud to allow for evacuation of the lubricating oil. This data was used in the modeling effort conducted in Reference 12. This was done with the shrouding at minimum and maximum conditions for the shrouds (Table 2). The data with the tare losses removed is shown in Figure 8. From this figure the shroud losses are much more for the case of large clearance and becomes much more efficient when the shrouding is pulled in close to the gear at the minimum radial and axial clearances. The data indicates that the radial clearance affects are more important than the axial clearance. The windage loss was reduced by over 50 percent by minimizing the radial and axial clearance as compared to the shrouds in the maximum radial and axial clearance configuration.

Also, during these tests data was taken from a thermocouple and static pressure probe on the outer circumference of the shroud at the mid point across the gear tooth face width. The thermocouple and static pressure probes enabled the shroud internal temperature and pressure to be tracked during the stepped increase of rotational speed of the gear-shaft assembly. The data from these two sensors are shown in Figures 9 and 10, respectively.

The data shown in Figure 9 was not a steady state value (thermally), but more an indicator of shroud pitch line velocity effects. Once again the minimum axial and radial clearance produced the lowest fling-off temperatures.

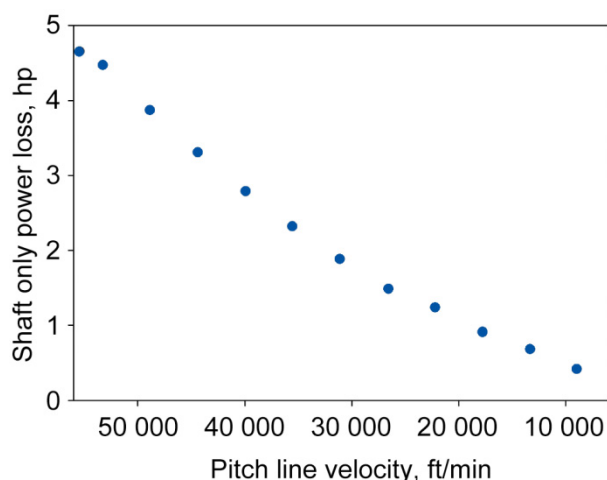


Figure 7.—Input shaft only results.

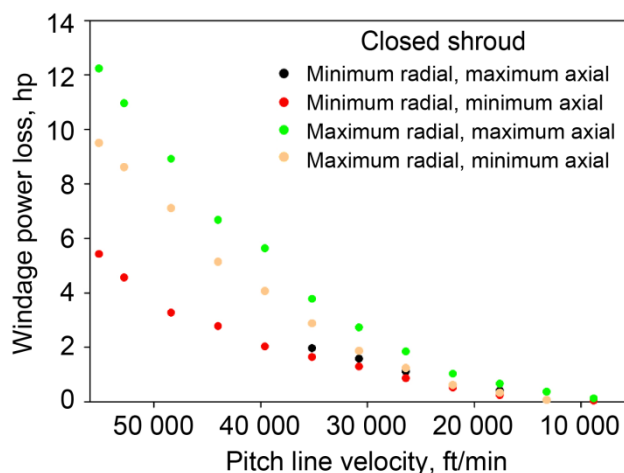


Figure 8.—Closed shroud windage power loss as a function of pitch line velocity.

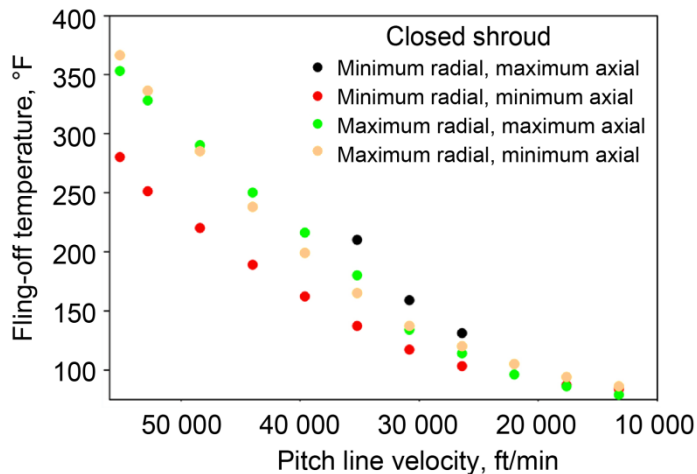


Figure 9.—Closed shroud fling-off temperature as a function of pitch line velocity.

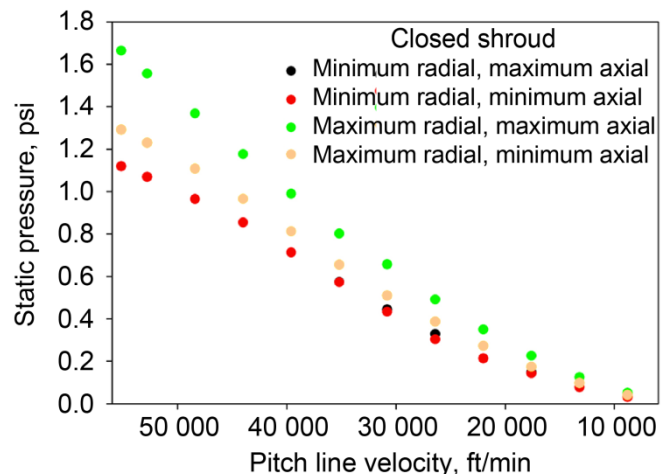


Figure 10.—Closed shroud static pressure as a function of pitch line velocity.

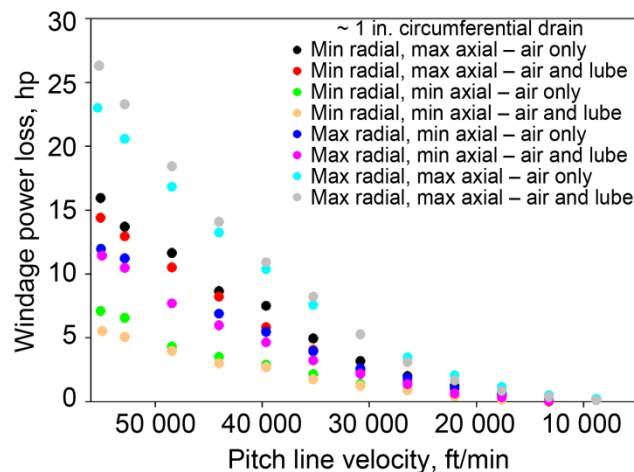


Figure 11.—Windage power loss for single gear with 1-in. circumferential drain, with and without lubricant present.

Finally in Figure 10 for the closed shroud data, the static pressure at the shroud wall is presented as a function of pitch line velocity. The minimum axial and radial clearance had a reduced static pressure when compared to the other arrangements.

4.2 Shroud With Drain Slot

Another series of tests were run with the single gear inside a shroud with a single drain slot, with and without the lubricating oil present. The tests were first conducted in air to ensure that no lubricant was inside the shroud, and then the lubricant would be turned on and the test sequence repeated. The lubricating jet temperature was maintained near the ambient conditions.

The data for the windage power loss is shown in Figure 11. The shrouding was positioned as shown in Table 2 for the minimum and maximum radial and axial positions. The lubricant, when present, was injected radially at the mid-face width location at a pressure of ~100 psi and with a flow rate of ~0.4 gpm. The lubricant then drains out of the shroud (1 in. rectangular slot) located at the bottom. As can be seen

from Figure 11, the minimum axial and radial shroud locations produced the lowest power loss. When compared to the maximum-maximum conditions (Table 2), the minimum clearances had at least a 4X decrease in power loss.

The fling-off temperature and static pressure data are shown in Figures 12 and 13 for the same tests as already discussed in Figure 11. For all cases the “air and lube” condition immediately followed the “air only” condition. The residual heat from the “air only” test resulted in an elevated initial fling-off temperature for the “air and lube” test. With the introduction of the relatively cooler lubricant (~room temperature) at the beginning of the test, the “air and lube” fling-off temperature exhibited a negative trend with increasing speed. However, the work on the air-lube mixture from the rotating gear and shaft assembly eventually reversed this trend in fling-off temperature as shown in this figure.

As mentioned in the earlier discussion, the temperature data shown was not a steady state value. Data was taken relatively quickly as the speed and torque data stabilized. For the data shown in Figures 12 and 13, the minimum radial, maximum axial clearance shrouding produced the lowest fling-off temperature and static pressure. The maximum radial clearances produced the highest fling-off temperatures and static pressures.

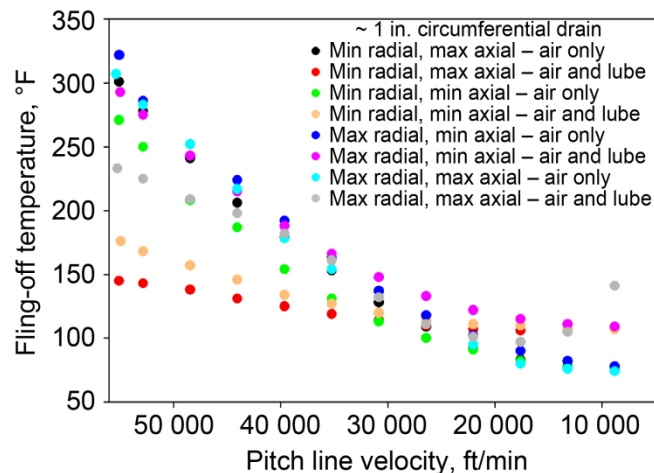


Figure 12.—Fling-off temperature for single gear, 1-in. circumferential drain, with and without lubricant present.

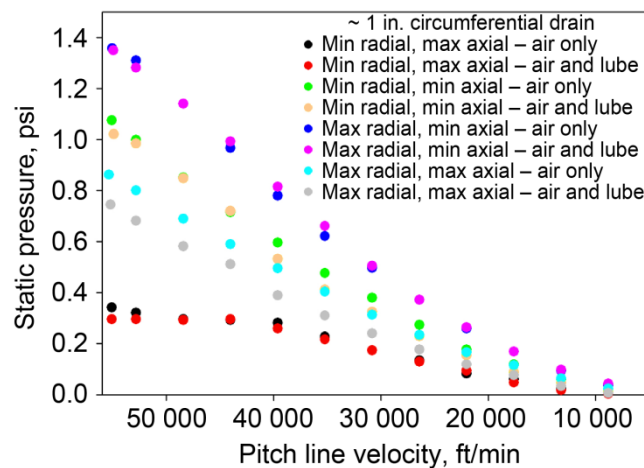


Figure 13.—Static pressure for single gear, 1-in. circumferential drain, with and without lubricant present.

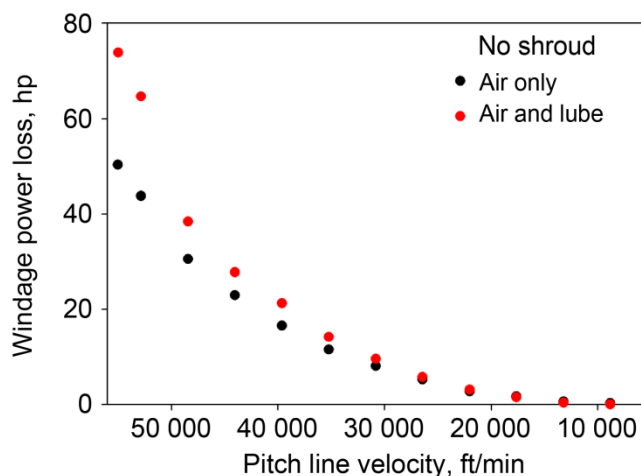


Figure 14.—No shroud windage power loss.

4.3 No Shroud Results

Finally the data from the same gear operating with and without lubricant flowing and without a shroud is presented. This arrangement produced the largest losses from a windage standpoint as compared to any of the other shrouded conditions discussed in this paper. The data for the no-shroud conditions is shown in Figure 14 for windage power loss. For the highest speed conditions the losses were 3 to 5 times greater than that of the worst performing shrouded conditions.

Conclusions

A study was conducted to determine the windage power loss from a single gear. The gear was operated from 8800 to 55000 ft/min (44.7 to 279.5 m/s) pitch line velocity. The gear was run in an enclosed shroud, a shroud with a single drain slot and completely without shrouds. For the shroud with a drain and no shroud conditions the gear was run with and without a lubricating jet present.

The following conclusions can be drawn from this preliminary study:

1. For very high pitch line velocities shrouding, even loosely fitting to the rotating gear is extremely beneficial in reducing windage power losses.
2. Minimum axial and radial clearance between the shrouding and the gear tips and gear sides produced the lowest windage power losses.
3. For the closed shroud tests (air only), the fling-off temperature and static pressure followed the same trends as the windage power loss.
4. For the data taken with drained shrouds, the minimum axial and radial clearance conditions produced the lowest windage power loss, fling-off temperatures, and lowest static pressure.

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